PB90-262833

Preliminary Selection of Refrigerants for Dual-Circuit and Lorenz Refrigerator/Freezers

(U.S.) Environmental Protection Agency, Research Triangle Park, NC

1990

U.S. Department of Commerce
National Technical Information Service
NTIS

PB90-262833

90-138.2

EPA/600/D-90/104

Preliminary Selection of Refrigerants for Dual-Circuit and Lorenz Refrigerator/Freezers

Jane C. Bare
Air and Energy Engineering Research Laboratory
Office of Research and Development
U. S. Environmental Protection Agency
Research Triangle Park, North Carolina 27711

For Presentation at the
83rd Annual Meeting of A&WMA
Pittsburgh, Pennsylvania
June 24-29, 1990

AEERL-P-647

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before comple

1. REPORT NO.	2.	3.
EPA/600/D-90/104		PB90-262833

4. TITLE AND SUBTITLE	5. REPORT DATE
Preliminary Selection of Refrigerants for Dual-Circuit and Lorenz Refrigerator/Freezers	6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S)	8. PERFORMING ORGANIZATION REPORT NO.
Jane C. Bare	
9. PERFORMING ORGANIZATION NAME AND ADDRESS	10. PROGRAM ELEMENT NO.
See Block 12	11. CONTRACT/GRANT NO.
12. SPONSORING AGENCY NAME AND ADDRESS	13. TYPE OF REPORT AND PERIOD COVERED
EPA, Office of Research and Development Air and Energy Engineering Research Laboratory Research Triangle Park, NC 27711	Published paper 14. SPONSORING AGENCY CODE EPA/600/13

15. SUPPLEMENTARY NOTES AEERL project officer is Jane C. Bare, Mail Drop 62B, 919/541-1528. For presentation at 83rd Annual Meeting, AWMA, Pittsburgh, PA, 6/24-29/90.

16. ABSTRACT The paper discusses the preliminary selection of refrigerants for dual-circuit and Lorenz refrigerator/freezers (RFs). Since RFs currently use a fully halogenated chlorofluorocarbon (CFC) as a refrigerant and are the most energy-consuming appliances in most households in the U.S., EPA is concerned about not only RFs' effects on stratospheric ozone depletion but also their impact on the greenhouse effect. Reductions in power consumption will slow the increase in carbon dioxide (CO_2) concentrations in the atmosphere by reducing the utility companies' need for capacity. CO_2 concentrations are believed to be directly related to impacts of the greenhouse effect. For these reasons, EPA believes that finding the most energy-efficient refrigerant(s) and RF design is important when eliminating fully halogenated CFCs. Two new configurations (the dual-circuit and Lorenz RFs) appear to provide energy savings of up to 20% over standard U.S. RF designs.

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U.S. DEPARTMENT OF COMMERCE
NATIONAL TECHNICAL
INFORMATION SERVICE
SPRINGFIELD, VA 22161

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Pollution Refrigerants Refrigerators Freezers Chlorohydrocarbons Fluorohydrocarbons Ozone Stratosphere Greenhouse Effect Carbon Dioxide	Pollution Control Stationary Sources Chlorofluorocarbons	13B 07B 13A 04A 07C

18. DISTRIBUTION STATEMENT	19. SECURITY CLASS *(This Report)*	21. NO. OF PAGES
Release to Public	Unclassified	16
	20. SECURITY CLASS *(This page)*	22. PRICE
	Unclassified	

EPA Form 2220-1 (9-73)

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INTRODUCTION

In the mid-70s, concern mounted over the depletion of stratospheric ozone by fully halogenated chlorofluorocarbons (CFCs). Ozone depletion could lead to increasing UV-radiation, which in turn would lead to increasing cataracts, skin cancers, immune system deficiencies, and other human and ecological problems. Additionally, global warming due to the greenhouse effect could affect plant and animal life including agricultural products and could cause severe flooding in many coastal areas. To prevent these potential disasters from stratospheric ozone depletion, an international agreement titled the Montreal Protocol was signed in 1987 to reduce consumption of CFCs and halons.[1]

Since refrigerator/freezers (RFs) currently use a fully halogenated CFC as a refrigerant and are the most energy-consumptive appliance in most households in the U.S.[2], EPA is concerned about not only the RFs' effects on stratospheric ozone depletion but also their impact on the greenhouse effect. Reductions in power consumption will slow the increase in carbon dioxide concentrations in the atmosphere by enabling the utility companies to have less capacity. For these reasons, EPA believes that finding the most energy-efficient refrigerant(s) and RF design is important when eliminating the fully halogenated CFCs.

The ideal RF should be safe, energy-efficient, and inexpensive. This paper reviews the RF design and refrigerant selection process which is necessary to design an energy-efficient RF which does not use fully halogenated CFCs.

SELECTION OF REFRIGERANT AND REFRIGERATOR DESIGN

Most current RFs in the U.S. market are 0.51 to 0.62 m^3 (18 to 22 ft^3) with automatic defrost. The typical U.S. design contains an evaporator, a compressor, a condenser, an expansion valve, and R-12 as the refrigerant. See Figure 1 for the current RF design. While maintaining the size and automatic defrost feature, future variations to this design may include: 1) dual-circuit systems with two independent circuits for the freezer and fresh food, or 2) two-evaporator systems such as the Lorenz RF with Non-Azeotropic Refrigerant Mixtures (NARMs). The refrigerant(s) of choice may include one or two single refrigerants, or a NARM.

In light of the recent concern about global warming due to the greenhouse effect, EPA is interested in increasing the energy efficiency of RFs while removing the CFCs from the RF which deplete stratospheric ozone. Energy efficiency is determined by the ratio of available cooling to the energy necessary to accomplish the cooling. This ratio may be estimated by using the ideal Coefficient of Performance (COP) (enthalpy change in evaporator divided by enthalpy change in compressor). The ideal COP may be calculated by using data taken from Pressure-Enthalpy diagrams or by using computer models which provide calculated enthalpy data using thermodynamic relationships tied to an

Equation of State (EOS).

The COPs were evaluated for the cycles of interest using the National Institute of Standards and Technology's (former National Bureau of Standard's) Cycle7 heat pump model.[3] Although the Cycle7 model was written as a heat pump model, at the time of this analysis, it was one of the few personal computer models which enabled the user to analyze vapor-compression cycles with many of the new alternative refrigerants. Cycle7 was run at typical RF temperatures in an attempt to most closely simulate a RF in this analysis.

Several built-in assumptions in this analysis are not consistent with "real life" RFs. First, the Cycle7 model is written to simulate heat pumps which do not have a heat exchanger in the compressor suction line. In reality, RFs are designed with this heat exchanger to provide at least 3 C° superheating into the vapor region, thus ensuring that no liquid will enter the compressor. Second, compared to heat pumps, RFs have comparatively longer lines attaching the evaporator and the condenser to the compressor. These lines are uninsulated and lead to superheating and subcooling which is unaccounted for in the model. Cycle7 also assumes perfect isentropic compression in the compressor, and perfect isenthalpic expansion in the expansion valve. This analysis assumed that there were no pressure drops in the condenser and evaporator. In reality, there are pressure drops in all lines in the system, various inefficiencies inherent to the compressor, and other frictional losses. Despite all of the above limitations, the selection of refrigerants using Cycle7 was later confirmed on a recently released RF computer model.

The Cycle7 model is based on the Carnahan-Starling-DeSantis (CSD) EOS which requires refrigerant-specific parameters. These parameters were determined by using saturated vapor pressure, vapor density, and liquid density data for each refrigerant. These parameters were provided as a data file in the Cycle7 model.

The first list of refrigerants chosen for modeling was limited to those refrigerants included in the Cycle7 database. This list was then reduced by considering: stratospheric ozone depletion potential, anticipated global warming potential, energy efficiency, operating conditions, flammability, toxicity, and availability. Table I lists the current status of these characteristics for many of the alternative refrigerants.

Although some power is used by the fans (approximately 9%), automatic defrost heaters (approximately 5%), and anti-sweat heaters (approximately 11%), the majority of the RF power requirements are in the compressor (approximately 74%).[4] The power required by the compressor depends on a number of factors, many of which relate to inefficiencies in the compressor, but the most important are suction and discharge pressures. One reason that inefficiencies occur is when the evaporating temperature for either compartment is lower than necessary for heat transfer.

Both the dual-circuit and the Lorenz RFs operate at a higher temperature in the fresh food evaporator, and thus present the potential for energy savings.

DUAL-CIRCUIT ANALYSES

A dual-circuit RF can consume less energy because two specially designed loops are used: one loop for cooling the freezer and the other for cooling the fresh food. Figure 2 illustrates a dual-circuit design. The fresh food evaporator may be run at higher temperatures/pressures, thus using less compression energy. An additional benefit may result if the optimum refrigerant is utilized for each loop.

The purpose of the dual-circuit analysis was to determine the optimum refrigerant(s) for each circuit and calculate the possible energy savings of using two circuits instead of one. In this analysis, the system was analyzed using:

Temperature of air entering condenser = 32 °C
Temperature of air leaving condenser = 40 ° C
Log Mean Temperature Difference = 5 °C

Freezer:
Temperature of air entering evaporator = -15 °C
Temperature of air leaving evaporator = -20 °C

Fresh food:
Temperature of air entering evaporator = 3 °C
Temperature of air leaving evaporator = -2 °C

For a dual-circuit system using pure refrigerants, these inputs result in a freezer evaporation temperature of -23 °C, and a fresh food evaporation temperature of -5 °C. The results of the dual-circuit models were compared to those of the traditional R-12 single-circuit system at the evaporation temperature equal to -23 °C.

The refrigerants, COPs, capacities, and suction and discharge pressures within the system are listed in Tables II and III for the freezer and fresh food sections, respectively.

Using the results from Cycle7, the refrigerants were screened for inadequate performance in the system. This was defined as: operating under vacuum (less than 101.4 kPa), operating with high pressures (greater than 1520 kPa), exceeding critical temperature in condenser, or operating with two phases in the compressor. R-13, R-14, and R-23 were eliminated for exceeding critical temperature in the condenser; R-22 and R-32 for operating with high pressure; and R-123 and R-124 for operating under vacuum. R-142b was operating under vacuum at freezer conditions, but operating within the range of acceptable pressures in the fresh food evaporator.

After this process of elimination, only three viable refrigerants in the fresh food section remained. They are, in order of largest COP: R-142b, R-152a, and R-134a. After elimination in the freezer section, there were only two viable refrigerants. They are in order of largest COP: R-152a and R-134a. The two refrigerants with the highest performance—R142b in the fresh food circuit and R-152a in the freezer circuit—were then analyzed in a two-circuit system vs. the traditional R-12 one-circuit system using a method of comparison similar to the method used by Pedersen.[5]

Pedersen shows that, for the assumptions of equal isentropic efficiency and motor efficiency for both systems, the power consumption of the compressor (N) may be calculated with the COP and the cooling capacity (Q).

If one circuit is used, the power consumption of the compressor (N_1) may be calculated with the COP of the freezer (COP freezer) and the cooling capacity of the freezer (Q freezer) and fresh food (Q fresh food), since evaporation is occurring at the freezer temperature. Thus, for a one-compressor system:

N_1 = (Q freezer + Q fresh food)/COP freezer

whereas, for a two compressor system:

N_2 = (Q freezer/COP freezer) + (Q fresh food/COP fresh food)

Assuming that the heat loads for the fresh food compartment and the freezer compartment are equal, the relative power consumption becomes only a function of COPs.

In this case, the equation for R-12 in a one-compressor system becomes:

N_1 = (0.5 + 0.5)/2.831 = 0.3532321

In a dual-circuit system with R-152a in the freezer and R-142b in the fresh food, the equation is:

N_2 = (0.5/2.981) + (0.5/4.819) = 0.2714849

The relative saving in compressor power consumption is:

$(N_1 - N_2)/N_1$ = 23%

Experimental work to confirm the model predictions will be conducted at an EPA-funded laboratory looking at the dual-circuit system.

LORENZ RF

In a 1975 International Institute of Refrigeration (IIR) paper, Lorenz and Meutzner claimed that a two-evaporator, two-subcooler RF had shown energy savings of up to 20%.[6] Figure 3 is the Lorenz RF design. Unfortunately, their analysis was based on using R-22/R-114, and R-114 is now considered a strong ozone depleting compound on the list of substances mentioned in the Montreal Protocol.[1] The purpose of this analysis was to determine the optimum NARMs for a Lorenz RF which do not include fully halogenated CFCs.

The Cycle7 model was run to simulate the Lorenz RF by assuming that the two evaporators and two subcoolers could be treated as a heat exchanger unit. Various combinations of refrigerants were tested in the Cycle7 model at 0.05 composition increments to estimate capacity, COP, and operating pressures for each NARM composition.

Using the results from Cycle7, the refrigerants were screened for adequate performance in the system as before. Inadequate performance was defined as: operating under vacuum (less than 101.4 kPa), operating with high pressures (greater than 1520 kPa), exceeding critical temperature in the condenser, or having two phases in the compressor.

The model results for the NARMs were compared for their COPs, volumetric capacities, compressor suction pressures, and compressor discharge pressures when evaporating at near atmospheric pressure in the heat exchanger unit. Based on these COPs, the top NARMs were chosen for future experimental work.

Experimental work to confirm the model's result will be conducted at EPA-funded laboratories looking at the Lorenz system.

CONCLUSIONS

Dual-circuit RFs have been shown to have a theoretical advantage over single-evaporator RFs, since evaporation occurs at two different pressures.[5] This process requires less compressor work for the higher temperature/pressure evaporation. Single refrigerants were analyzed to determine the optimum pure refrigerant in each of the two separate freezer and fresh-food loops. R-152a was determined to be the optimum refrigerant in the freezer, and R-142b was determined to be the optimum refrigerant in the fresh-food section. With the assumptions made, theoretical energy savings of up to 23% of compressor power are possible. Experimental work to confirm the model predictions will be conducted at an EPA-funded laboratory looking at the dual-circuit system.

The Lorenz two-evaporator/two-subcooler design RF, presented at the IIR Conference in 1975, also uses a two-evaporator RF, and

has shown energy savings of up to 20% when using a NARM.[6] The first step in optimizing this design for use in the U.S. market is the selection of NARMs for testing. These NARMS will be tested in two-evaporator RFs at EPA funded laboratories.

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TABLE I Potential refrigerant substitutes for CFCs.

Refrigerant	Global Warming Potential (a)	Ozone Depletion Potential (b)	Commercial Status	Status of Toxicological Testing (Feb. 1990)	Flammability
HFC-152a	33	0	in production	tested	mild
HCFC-123	21	0.013-0.027	in pilot production	in testing	none
HFC-32	(c)	0	was produced in limited quantities	limited testing	very mild
HCFC-124	110	0.013-0.030	(d)	in testing	none
HCFC-22	420	0.032-0.071	in production	tested	essentially none
HFC-134a	320	0	in pilot production	in testing	none
HCFC-142b	420	0.035-0.077	in production	tested	mild
HFC-125	630	0	(d)	preliminary testing	none
HFC-143a	900	0	(d)	Preliminary testing	(d)

Sources and Notes:

(a) Estimated by Global Change Division, U.S.EPA, based on WMO, NASA, and EPA sources (see references 7-10). These GWP values are relative to carbon dioxide, which has been assigned a value of 1. These GWPs are assessed on a mass basis over an infinite time period. The three elements used to assess GWPs, namely radiative forcing, atmospheric lifetime, and temporal integrating period, are still being investigated, so these data are subject to change.

(b) D. A. Fisher, et al., "Relative Effects on Stratospheric Ozone of Halogenated Methanes and Ethanes of Social and Industrial Interest," in UNEP/WMO Scientific Assessment of Stratospheric Ozone: 1989 Appendix: AFEAS Report (September 5, 1989), Table IV. Data reflect the range of estimates from four different 2-dimensional models. Data are relative to CFC-11, which is assigned a value of 1.

(c) Not computed, but probably low.

(d) Information not available.

TABLE II Freezer replacements for dual-circuit RFs.

Evaporating Temp = -23 °C (-30 °F)

Refrigerant	COP	Suction Pressure (kPa)	Discharge Pressure (kPa)
R-11	3.162	14	181
R-12	2.831	134	989
R-13 [a]	--	--	--
R-13B1 [c]	2.413	408	2290
R-14 [a]	--	--	--
R-22 [c]	2.918	217	1540
R-23 [a]	--	--	--
R-32 [c]	2.940	361	2432
R-113 [d] [b]	2.974	4	84
R-114 [d] [b]	2.649	32	356
R-123 [b]	3.013	11	167
R-124 [b]	2.768	67	619
R-125 [c]	2.313	302	2047
R-134a	2.771	116	1060
R-142b [b]	3.039	59	554
R-143a [c]	2.473	279	1814
R-152a	2.981	109	934
Propane	2.653	216	1421

Conditions for Elimination:

[a] Critical temperature exceeded in condenser
[b] Operating under vacuum (<101.3 kPa)
[c] Operating with high pressure (>1520 kPa)
[d] Saturated evaporator outlet results in two-phase compressor discharge

TABLE III Fresh food replacements for dual-circuit RFs.

Evaporating Temperature = -5°C (-21 °F)

Refrigerant	COP	Suction Pressure (kPa)	Discharge Pressure (kPa)
R-11 (b)	4.978	32	183
R-12	4.561	260	99
R-13 (a)	--	--	--
R-13B1 (c)	3.961	723	239
R-14 (a)	--	--	--
R-22 (c)	4.628	420	1566
R-23	--	--	--
R-32 (c)	4.635	692	2497
R-113 (d) (b)	4.791	12	84
R-114 (d) (b)	4.432	72	356
R-123 (d) (b)	4.820	27	167
R-124 (d)	4.533	139	619
R-125 (c)	3.865	571	2058
R-134a	4.472	243	1065
R-142b	4.819	122	556
R-143a (c)	4.090	519	1823
R-152a	4.736	223	994
Propane	4.321	405	1430

Conditions for Elimination:

(a) Critical temperature exceeded in condenser
(b) Operating under vacuum (<101.3 kPa)
(c) Operating with high pressure (>1520 kPa)
(d) Saturated evaporator outlet results in two-phase compressor discharge

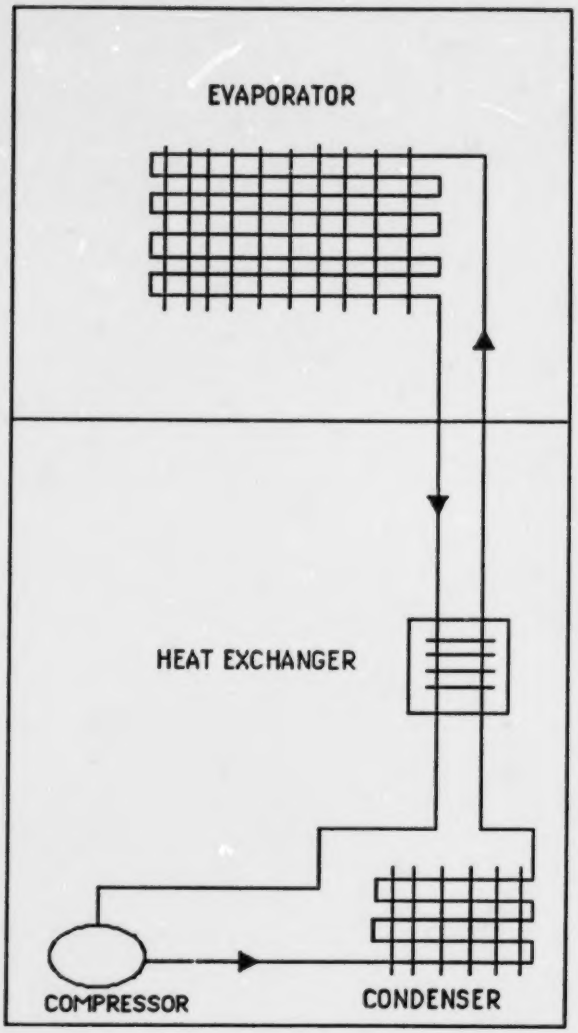

FIGURE 1:

TRADITIONAL REFRIGERATOR/FREEZER

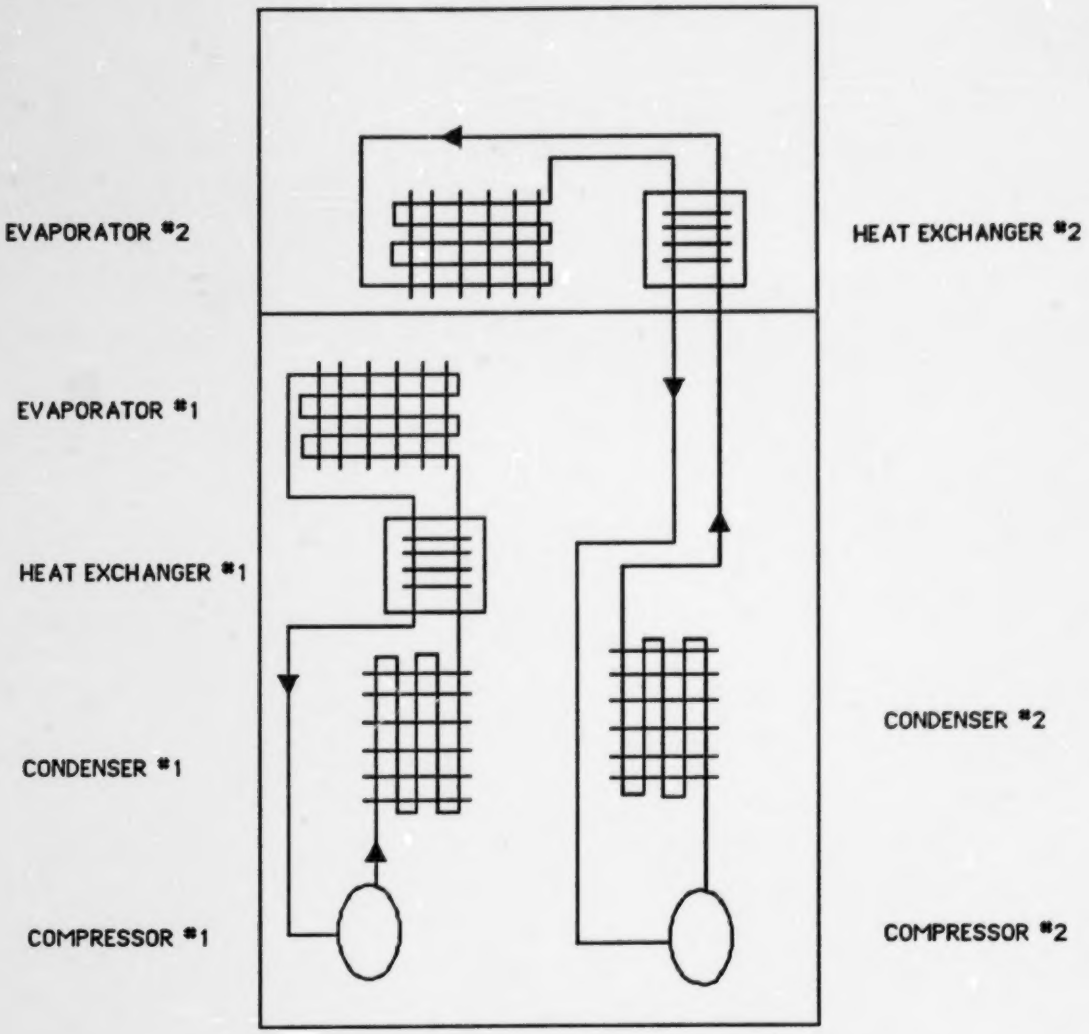

FIGURE 2:

DUAL CIRCUIT REFRIGERATOR/FREEZER

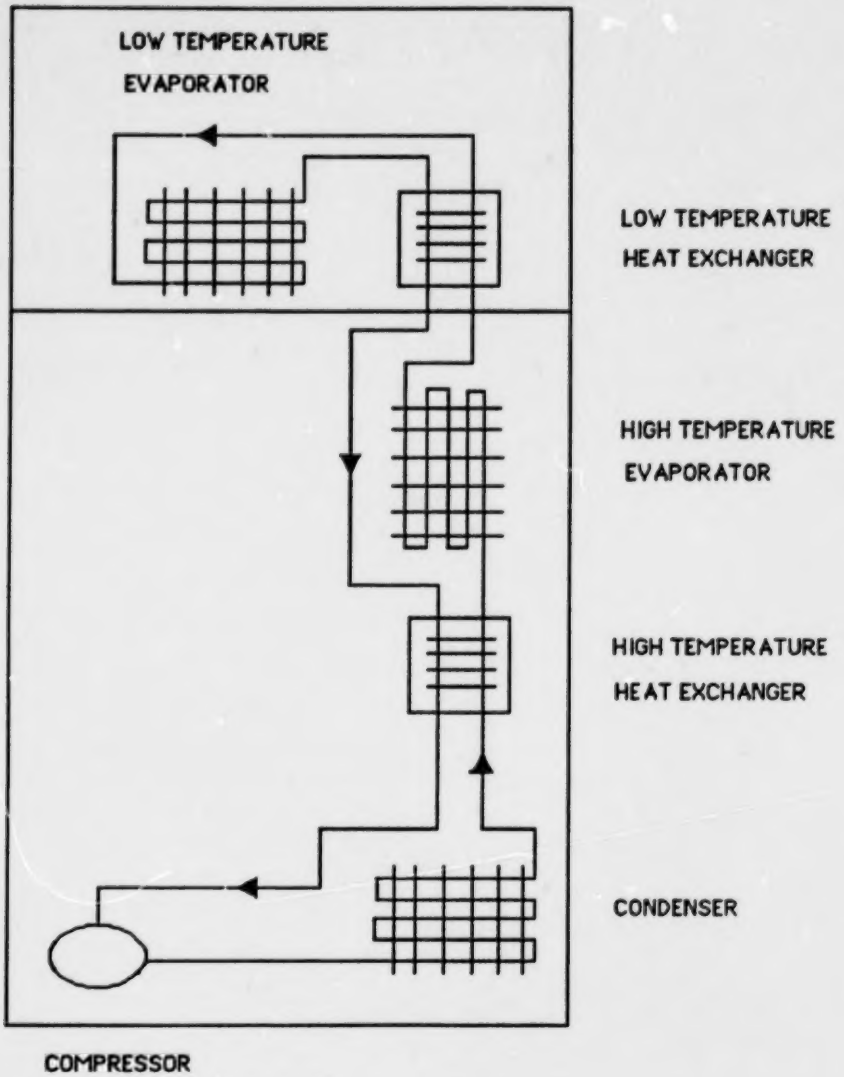

FIGURE 3:

LORENZ REFRIGERATOR/FREEZER